Cryogenic Thermomechanical Compressor

Simonenko Iu.M.[1], Chygrin A.A.[2], and Kostenko Ye.V.[1]

[1]National University of Technology, Odessa, Ukraine; [2]Cryoin Engineering Ltd., Odessa, Ukraine

Abstract. The purpose of this work is to create a compact supercharger to provide circulation of protective gas medium in a closed circuit. It was proposed to use a thermomechanical compressor to achieve this purpose. The operating principle of such devices is to change cyclically the temperature of the working medium in contact with warm and cold sources. Heating and cooling are carried out sequentially, pushing a part of gas through the regenerator by means of a displacer. The energy consumption for piston displacement is lower by an order of magnitude than that in conventional compressors. This makes it possible to use a seamless displacer movement mechanism. There can be two designs, both with one of the heat carriers close to ambient temperature. In a high-temperature thermomechanical compressor, the temperature is usually does not exceed 800 K. In the second type compressor, by reducing the absolute temperature of the cold "source" it is possible to achieve a high degree of compression at a relatively small temperature difference. The most significant result of the work is the design of the small-sized thermos-compressor that ensures a moderate degree of gas compression. The significance of the achieved results is shown in the compactness and tightness of the prototype for the use as an alternative to traditional machines in the field of inert gases production. The proposed technical solutions were tested during bench tests of the thermomechanical compressor. The experimental dependences were obtained of the flow rate characteristics on temperature mode, discharge pressure and cycle period.
Keywords: thermomechanical compressor, regenerator, inert gases, heat transfer, refrigerant.

DOI: https://doi.org/10.52254/1857-0070.2023.2-58.13
UDC: 621.56

Compresor termomecanic criogenic

[1]Simonenko Iu.M., [2]Cigrin A.A., [1]Kostenko Ye.V.

[1]Universitatea Națională tehnologică din Odesa, [2]Cryoin Engineering Ltd.
Odesa, Ucraina

Rezumat. Scopul lucrării este de a crea o schemă a unui compresor compact care să asigure circulația unui mediu gazos protector într-un circuit închis. Echipamentul criogenic se caracterizează prin condiții specifice de funcționare: presiune mare de funcționare de 10 ... 15 MPa, raport de compresie moderat de 1,2 ... 1,4 și productivitate relativ scăzută (mai puțin de 1 m3/h în condiții de aspirație). Prin urmare, utilizarea mașinilor voluminoase cu piston și diafragmă de tip tradițional pentru circulație nu este întotdeauna rațională. Pentru atingerea acestui scop au fost rezolvate următoarele sarcini: 1) a fost dezvoltat și testat un termocompresor; 2) se propune utilizarea unui drive magnetic; 3) a fost selectat un agent frigorific intermediar; 4) aplicarera procesului intern de recuperare a căldurii. Într-un termocompresor cu temperatură înaltă, temperatura este limitată de proprietățile de rezistență ale materialelor de construcție și, de obicei, nu depășește 800 K. Cel mai semnificativ rezultat al lucrării este dezvoltarea unui termocompresor de dimensiuni mici, care asigură un grad moderat de compresie a gazului ca alternativă la mașinile tradiționale din domeniul producției de gaze inerte. Semnificația rezultatelor obținute se manifestă prin faptul că au făcut posibilă realizarea unui prototip compact și etanș al unui termocompresor cu recuperare internă a căldurii, în care ansamblul pressepul a fost înlocuit cu un antrenament pe bază de magneți permanenți; utilizarea unui agent frigorific intermediar sigur - cripton - vă permite să excludeți tranziția de fază (condensare sau înghețare) a componentelor amestecului pompat. Soluțiile tehnice propuse au fost testate în procesul de teste pe banc.
***Cuvinte-cheie*:** compressor termomecanic, regenerator, gaze inerte, transfer termic, agent frigorific.

Криогенный термомеханический компрессор

[1]Симоненко Ю.М., [2]Чигрин А.А., [1]Костенко Е.В.
[1]Одесский национальный технологический университет; [2]ООО «Криоин Инжиниринг»
Одесса, Украина

Аннотация. Цель работы – создание схемы компактного нагнетателя для обеспечения циркуляции защитной газовой среды в замкнутом контуре. Для криогенной техники характерны специфические условия эксплуатации: высокое рабочее давление 10…15 МПа, умеренная степень сжатия 1,2…1,4 и относительно небольшая производительность (менее 1м^3/ч при условиях всасывания). Поэтому использование для циркуляции громоздких поршневых и диафрагменных машин традиционного типа не всегда рационально. Для достижения поставленной цели были решены следующие задачи: 1) разработан и испытан термокомпрессор; 2) предложено использовать магнитный привод; 3) осуществлен выбор промежуточного хладагента; 4) применение процесса внутренней регенерации тепла. Принцип работы таких устройств заключается в цикличном изменении температуры рабочего тела при контакте с теплым и холодным источниками. Для повышения экономичности нагрев и охлаждение осуществляют последовательно, переталкивая порцию газа через регенератор при помощи вытеснителя. Затраты энергии на перемещение поршня на порядок меньше, чем в компрессорах традиционного типа. Возможны две конструкции, в каждой из которых один из теплоносителей близок к температуре окружающей среды. В высокотемпературном термокомпрессоре температура ограничивается прочностными свойствами материалов конструкции и обычно не превышает 800 К. В компрессоре второго типа за счет уменьшения абсолютной температуры холодного «источника» можно реализовать высокую степень сжатия при относительно небольшой разности температур. Наиболее существенным результатом работы является разработка малогабаритного термокомпрессора, обеспечивающего умеренную степень сжатия газа, в качестве альтернативы машинам традиционного типа в области производства инертных газов. Значимость достигнутых результатов проявляется том, что они позволили создать компактный и герметичный прототип термокомпрессора с внутренней регенерацией тепла, в котором узел сальника заменен приводом на основе постоянных магнитов; использование безопасного промежуточного хладагента – криптона – позволяет исключить фазовый переход (конденсация или замерзание) компонентов перекачиваемой смеси. Предложенные технические решения апробированы в процессе стендовых испытаний.

Ключевые слова: термомеханический компрессор, регенератор, инертные газы, теплообмен, хладагент.

INTRODUCTION

It is known that gas pressure can be increased either by reducing its volume or by increasing its temperature. The latter option is implemented in so-called thermos-compressors. The simplest thermal pumps appeared several centuries ago and were practically displaced as piston engines improved.

More efficient compressors with thermal drives have appeared over the past few decades. This is achieved by removing heat input and output locations and introducing internal regeneration. These gas displacement processes were first implemented by V. Bush in 1939 [1].

A natural step in the improvement of thermos-compressors was the separation of the hot and cold parts of the working volume, which alternately came into contact with a portion of gas due to a movable displacer. The counter-pressure acting on the displacer was determined only by the hydraulic resistance of the system. As for the displacer, only a small amount of power was required to drive it. However, even after such modernization, thermos-compressors remained The peculiarity of real devices is the consumption of thermal energy for gas compression and mechanical energy for driving the thermomechanical compressor [2-4]. Moreover, this work requires significantly less energy for gas compression.

Such devices are mainly used in gasifiers and autoclaves.

Their main drawbacks were their inertia and significant heat losses. In addition to changing the parameters of the gas itself, the heat-intensive walls of the working chamber were also forced to heat up and cool down.

highly inertial and low-productivity devices [5-20].

As previously noted, the acceleration of heat exchange processes and the increase in cycle dynamics were achieved through internal heat regeneration. Simplified diagrams of the main layout options for the compressor are shown in Fig. 1.

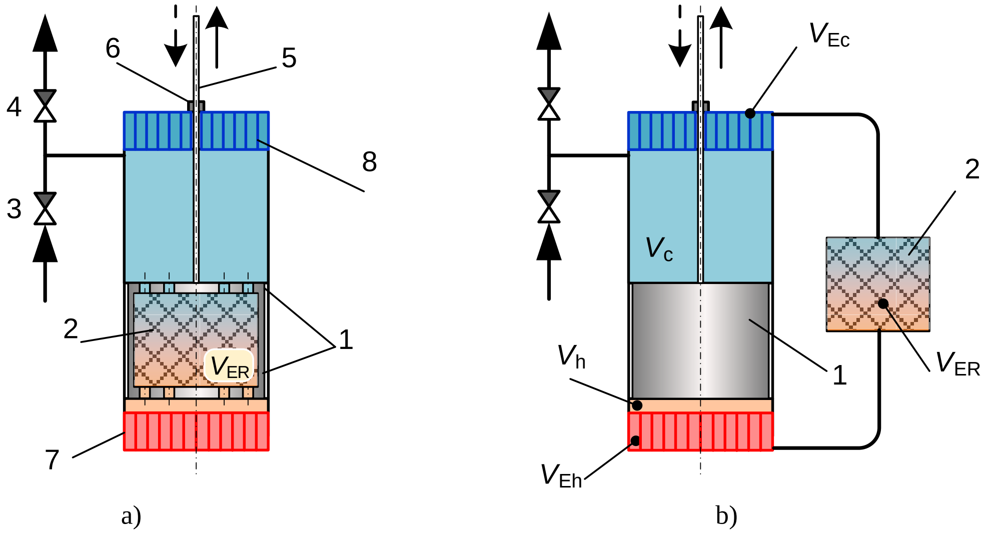


Figure 1. Schemes of thermocompressors: (a) – with the built-in regenerator in the displacer; (b) – with the external regenerator; 1 – the displacer; 2 – the regenerator; 3, 4 – the suction and discharge valves; 5 – the displacer rod; 6 – the gasket; 7 – the heater; 8 – the cooler; V_h and V_c – the operating cylinder warm and cold cavities' volumes; V_{Eh} and V_{Ec} – dead volumes of the heater and the cooler.

I. PRINCIPLE OF OPERATION AND ENERGY SUPPLY OPTIONS

The sequence of individual phases of the cycle is shown in Fig. 2, which illustrates the processes in the cold and hot chambers on the "volume-pressure" diagrams. As the displacer moves, the volumes of the cold V_c and hot V_h chambers cyclically change from the maximum value V_W to 0. At the same time, the sum of the volumes V_h and V_c remains constant throughout the cycle. The values V_{Ec} and V_{Eh} indicate the volumes of the heat exchange devices associated with the working chamber. The sum of the volumes of the cooler, heater, and regenerator forms the dead volume, as shown in formula

$$V_E = V_{Ec} + V_{Eh} + V_{ER} = \text{const.} \quad (1).$$

When designing heat exchangers, it is important to work for minimizing the dead volumes, as increasing the gas volumes of the heat exchangers in relation to the cylinder's working volume V_W reduces the compressor's productivity.

Depending on the position of the displacer, one cycle of the thermos-compressor can be divided into four separate processes [2, 3].

I-II - Pressure increase process. Displacer 1 is in the extreme lower position, and the volume of the hot chamber is zero. As the plunger moves, gas begins to be pushed from volume V_c to volume V_h through the regenerator, and its temperature increases. As a result, the pressure in the working cylinder begins to rise. One of the intermediate states is denoted by point x. Thermal compression continues until at point II the pressure reaches the value of the compression pressure P_D (Fig. 3).

II-III - Discharge process. The displacer continues to move up. Part of the gas, as before, flows into the hot chamber. The second part, through the open valve 4, enters the compressor line at pressure P_D.

III-IV – Pressure reduction process. At the beginning of the reverse (downward) stroke of the displacer, at point III, the compressor valve 4 closes. The compressed gas remaining in the hot space is pushed back through the regenerator to the cold space V_c. As a result of this cooling, the pressure in the cylinder drops to P_S (Fig. 3).

IV-I – Suction process. When the pressure is P_S, the suction valve 3 opens. As a result of the pressure drop and cooling of the gas, the cylinder begins to fill with a fresh portion of gas. Suction ends when the displacer reaches the lowest point (fragment I). The cycle then repeats. In the second option (Fig. 4a), temperature of the hot chamber T_h is ensured by a flow of air or flowing water, while the temperature level T_c is lowered to cryogenic temperatures. The θ value depends more on T_c. For example, if the θ value in a low-temperature compressor needs to be increased from 1.5 to 2.5, temperature T_c needs to be changed from 200 to 125 K, then for the same θ increase in a high-temperature compressor, an increase in T_h from 450 to 750 K is required. Naturally, the energy consumption for cryostatization of a low-temperature compressor exceeds the energy consumption for driving a high-temperature one. However, in some advanced technologies, there are waste streams of cryogenic liquid vapors, which can be used to activate low-temperature compressors designed for circulating high-pressure gas media. In this case, with the availability of cold streams, energy supply issues for cryogenic compressors become secondary.

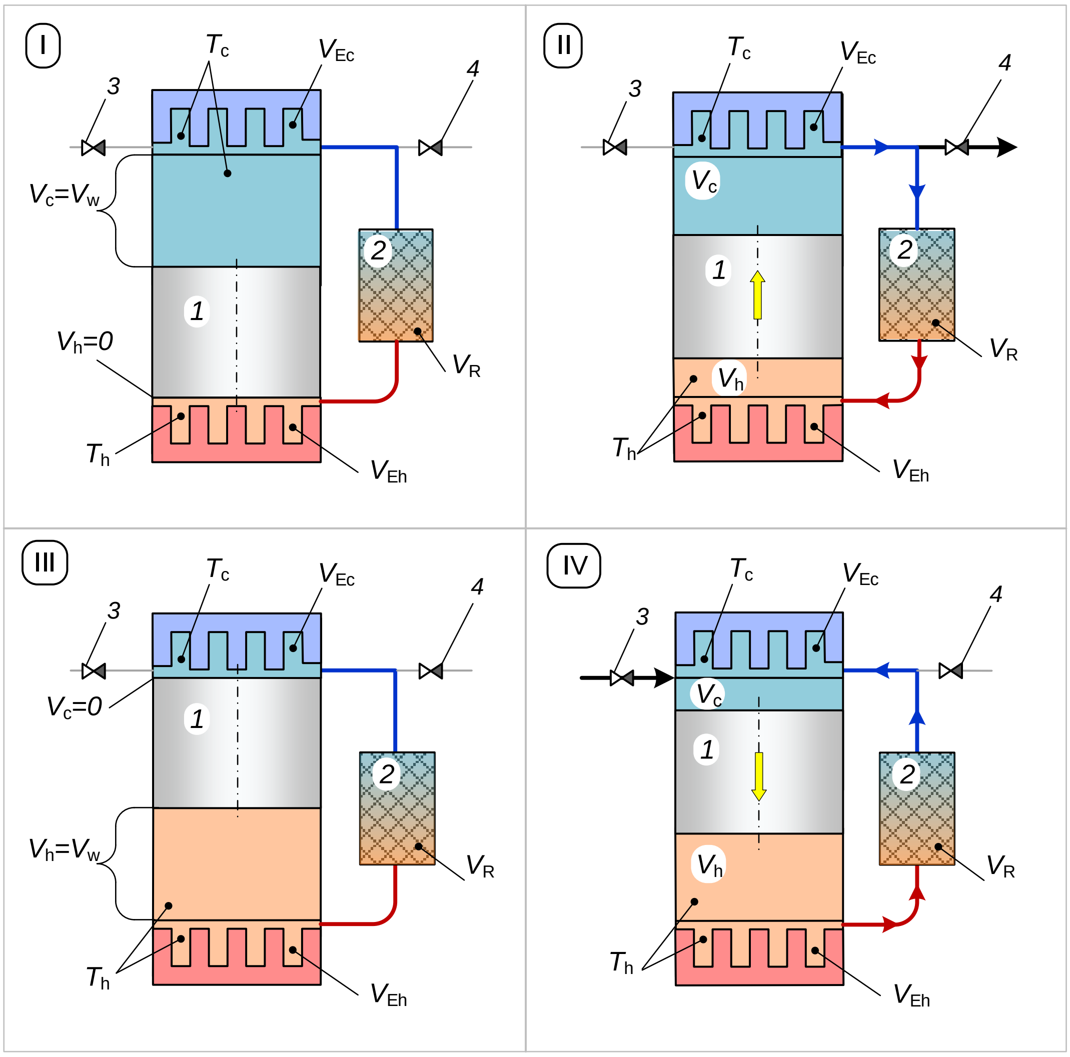


Fig. 2. The operation principle of the thermocompressor with an external location of the regenerator: 1…4 correspond to Fig. 1; V_W – the working volume of the cylinder, equal to the multiplication of the displacer motion 1 and its cross-sectional area; V_R, V_{Ec} and V_{Eh} –the hydraulic volumes of the regenerator, cooler and heater; V_c and V_h – the current volumes of the working cavity; T_c and T_h – the average gas temperatures in cold and warm cavities.

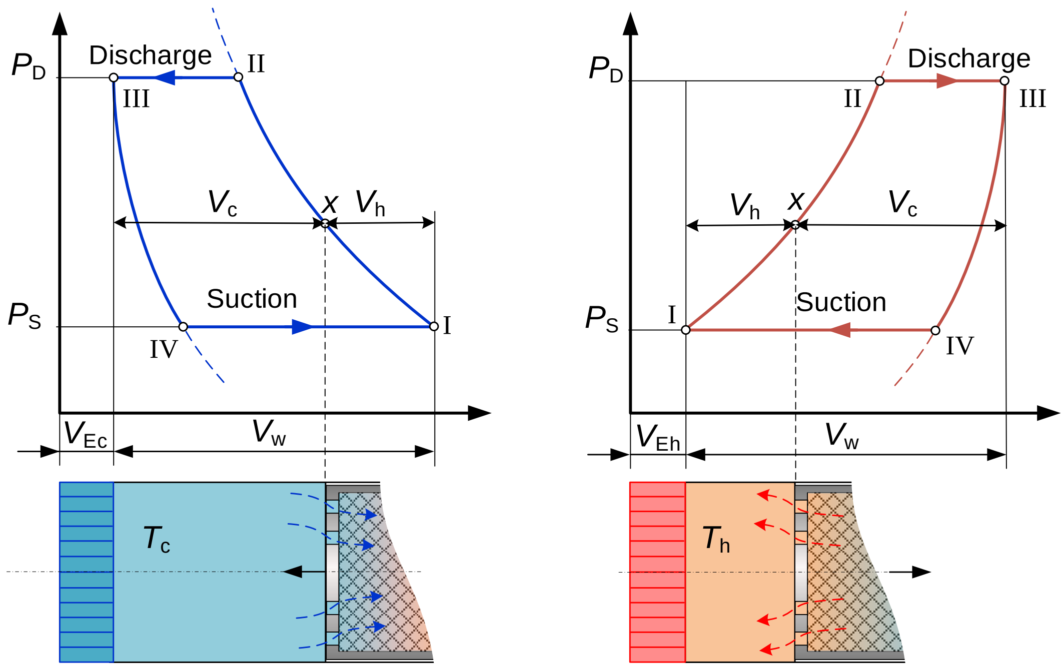


Fig. 3. The operating cycle in the diagram *P-V*: cold (a) and (b) warm cavities.

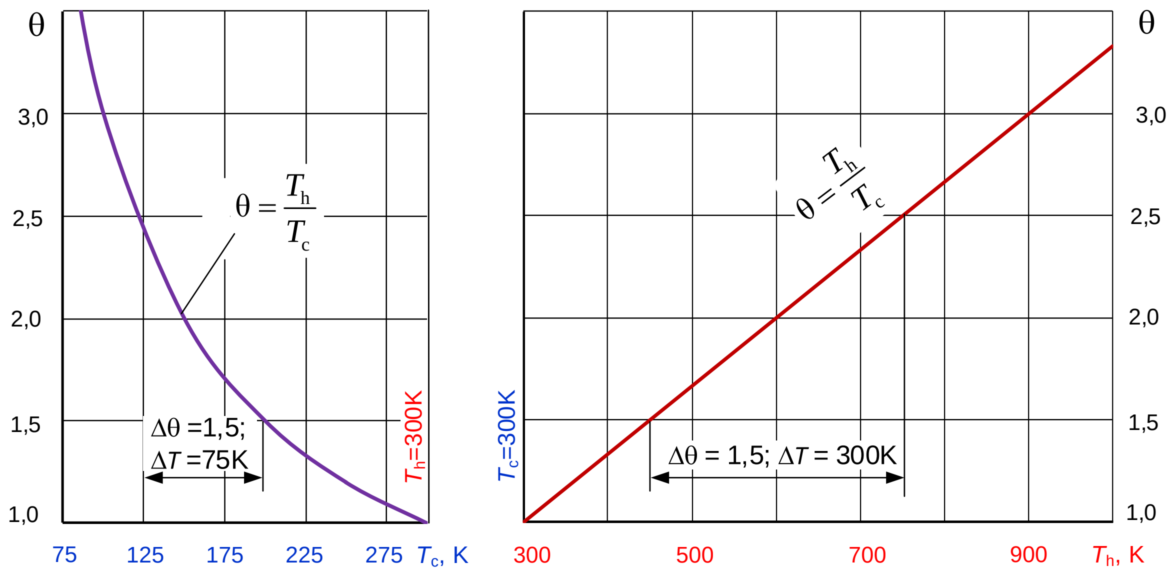


Fig. 4. The dependence between the ratio of operating temperatures θ for the low-temperature (a) and high-temperature (b) compressor at the temperature of one of the sources, close to ambient conditions (*T* = 300 K).

The presence of remaining gas in the heat exchange apparatus volumes V_{Ec}, V_{Eh}, and V_{ER} adversely affects the compressor's performance.

$$a = \frac{V_{Ec} + V_{Eh} + V_{ER}}{V_W}. \qquad (2)$$

However, excessive reduction of this parameter leads to a reduction in the heat exchange surface areas. Therefore, in real models, the relative dead volume usually fluctuates in the range of a = 0.5...1.0. The most important operating parameter of a thermo-mechanical compressor is the ratio of absolute temperatures of the working fluid in the hot and cold chambers.

When designing the heat exchangers, one should strive to minimize the total dead volume relative to the cylinder's working volume, as shown in formula (2).

$$\theta = \frac{T_h}{T_c}. \qquad (3)$$

Two temperature level options are possible, both assuming the use of one of the heat carriers close to the level of the ambient environment. In a high-temperature thermo-compressor, an electric heater is usually used as the second heat source. In this case, temperature T_h is limited by the strength properties of the structural materials and usually does not exceed 800…1000K (Fig. 4b).

II. FEATURES OF OPERATION OF A CRYOGENIC THERMOCOMPRESSOR IN THE PRESENCE OF HIGH-BOILING COMPONENTS IN THE PRODUCT

When pumping substances with a relatively high phase transition temperature, it is necessary to limit the temperature of the cold chamber T_c. Condensation or freezing of a component in a low-temperature thermos-compressor will block the regeneration process and lead to a change in the composition of the pumped mixture. The probability of this phenomenon increases at high pressures. Let's consider an example of pumping a mixture based on helium containing $y = 2\%$ tetrafluoromethane (CF_4; R14). The temperature of condensation of the high-boiling component is determined by its partial pressure in the mixture. In accordance with Dalton's law:

$$P_{\mathrm{R14}} = P_{\Sigma} \cdot y_{\mathrm{R14}} \approx P_{\mathrm{D}} \cdot y_{\mathrm{R14}}, \qquad (4)$$

where P_{Σ} - is the maximum pressure of the mixture in the cylinder determined by the pumping conditions P_D. The dependence of the condensation temperature of R14 on the specified concentration is presented in Table 1.

The calculated temperature in the cold cavity (last row in table 1) is chosen several degrees higher than the possible level of R14 condensation in the mixture. Such a step is justified in ensuring the prevention of phase transition in case of deviations from operational parameters, for example, an increase in the concentration $y_{\mathrm{R14}} > 2\%$ or a decrease in the load on heat exchangers due to the reduced performance.

Table 1.

Condensation temperature of tetrafluoromethane at different pressures of He-R14 mixture.

Maximum mixture pressure (discharge pressure) $P_{\Sigma} \approx P_D$, MPa	5	10	15	20
Partial pressure of tetrafluoromethane in a 2% mixture P_{R14}, MPa (4)	0,1	0,2	0,3	0,4
The temperature of tetrafluoromethane condensation at T_{R14}, K.	44,9	55,6	62,6	168,1
Allowable temperature in the cold box (with margin) T_c, K	50	61	68	173

When using nitrogen vapor as an external refrigerant, stabilizing the temperature T_c becomes relevant. Due to the significant temperature difference between N_2 and the mixture in the working chamber, local cooling of the heat exchanger sections is possible. Technical solutions based on regulating the refrigerant flow consumption depending on temperature T_c are fairly inertial. They do not prevent the occurrence of R14 refrigerant condensation and even its freezing at T < 90 K.

We have implemented a heat removal scheme using an intermediate substance (Fig. 5a). In the example under consideration, krypton and methane are suitable as intermediate refrigerants (Fig. 5b). As can be seen from the graphs, at allowable temperatures in the cold chamber T_c = 145...168 K, krypton is preferred because it is safe and at the same temperatures its boiling pressure is by 30...40% lower than in the case of CH_4.

III. IMPLEMENTATION OF THE ENGINEERING SOLUTION

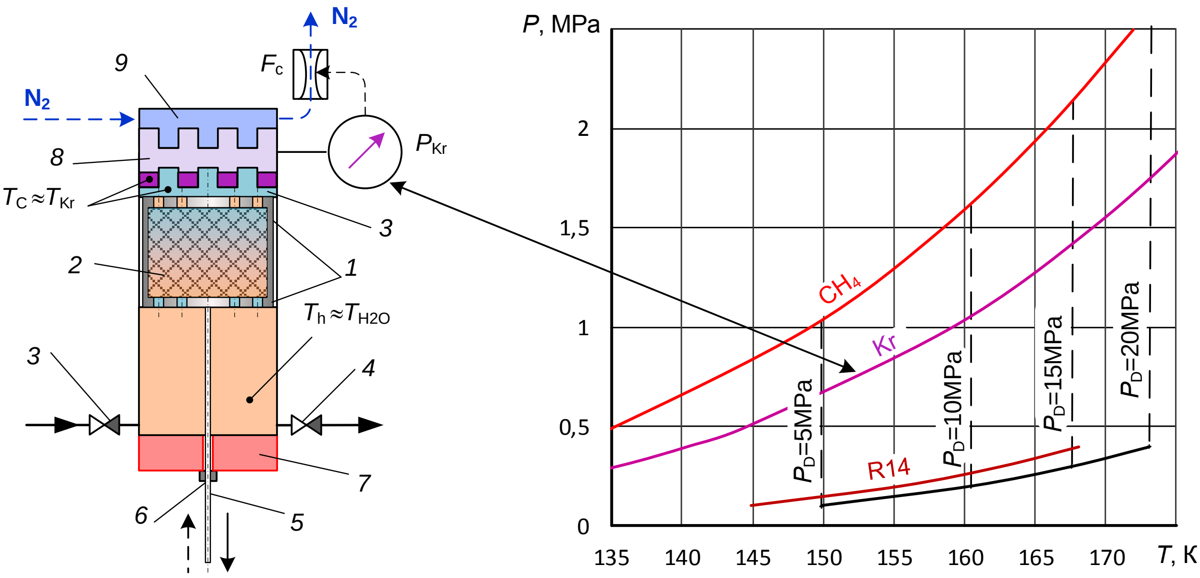
A prototype of a low-temperature thermo-mechanical compressor was manufactured and tested. A mechanism without a seal was used to drive the displacer (Fig. 6).

The transmission of motion through the hermetic wall was achieved using a balanced pair of neodymium Nd magnets. The total static force of the linear transmission was more than 100 N.

This value guaranteed the displacement of the displacer at a frequency of $\upsilon < 2\ \mathrm{c}^{-1}$, taking into account its mass, inertial forces, and piston ring friction. The movable support 3, which contains 12 magnets, surrounds the external wall of the stationary hermetic channel 1. In the photo (Fig. 6c), the cover of the support is removed, and only six magnets are visible.

The same number of magnets is symmetrically located on the other side of the cylindrical channel (Fig. 6b).

External magnets are in contact with the magnets 5 of the internal block. Assembly 2-3-4 performs reciprocating motion due to a pneumatic cylinder attached to links 8 of the support and controlled by a solenoid valve SV (Fig. 7). The characteristics of the model and the results of experimental studies are presented in table 2.



a) b)

Figure 5. Scheme of low-temperature thermocompressor with heat exhaustion by means of the intermediate refrigerant (krypton). Designations 1...6 - correspond to Fig. 1; 7 – water "heater"; 8 – cooler filled with two-phase krypton; 9 – external refrigerant (N_2); P_{Kr} – pressure in the intermediate refrigerant cavity; F_c – external refrigerant flow regulator; recommended pressure (b) in the cavity of the intermediate refrigerant, depending on the discharge pressure, at which the phase transition of R14 is excluded at its concentration in the mixture $y_{R14} = 2\%$ (table 1).

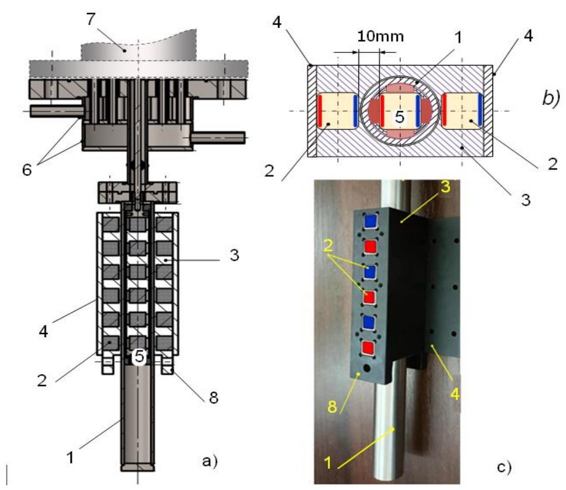


Fig. 6. Magnetic-mechanical drive device: a) – longitudinal section; b) – cross section; c) – view with the caliper cover removed; 1 – stationary sealed wall connected to the volume of the working chamber; 2 – magnets of the external unit; 3 – movable support for the external unit magnets; 4 – caliper cover; 5 – magnets of the internal unit; 6 – cooling water collectors; 7 – working cylinder; 8 – mechanical drive rods.

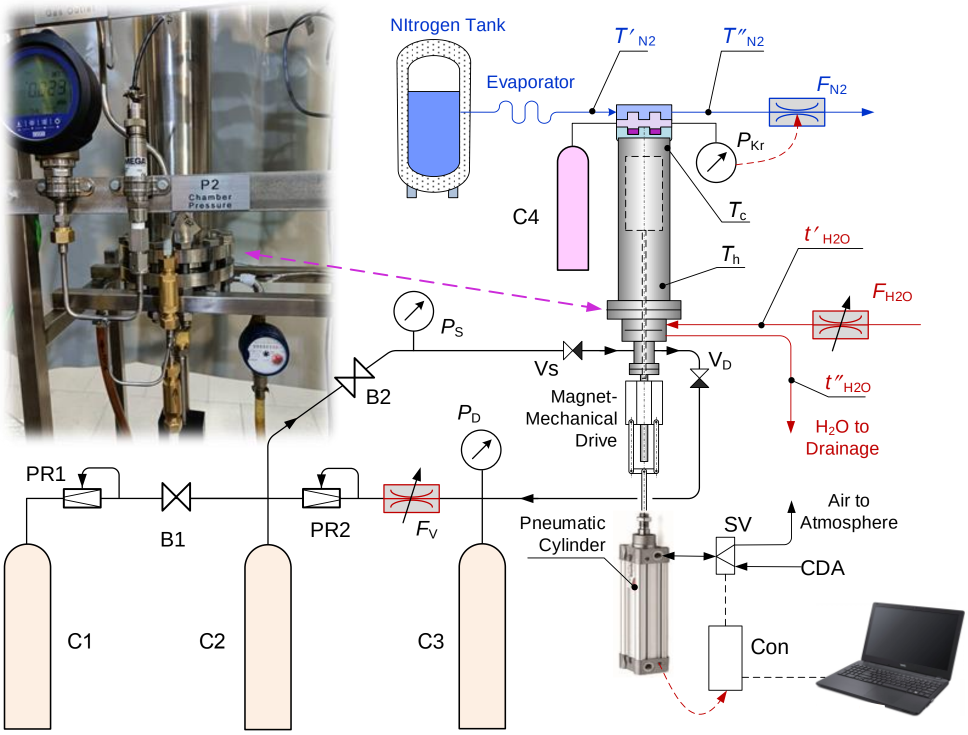


Figure 7. Simplified scheme of the test bench: C1 – cylinder with pumped mixture; C2 and C3 – buffers on the suction and discharge lines; C4 – cylinder with the intermediate refrigerant (Kr); PR1 – pressure regulators "AFTER itself"; PR2 – pressure regulator "BEFORE itself"; B1, B2 – valves; SV - solenoid pneumatic valve; Con – controller; CDA – pneumatic network; VS and VD – suction and discharge valves; FV – circulating flow meter; FH2O – "heating" water flow meter; FN2 – external refrigerant flow regulator (N2); PS and PD – suction and discharge manometers; PKr – pressure in the intermediate refrigerant cavity; t and T – temperature sensors.

Table 2.

Geometric parameters of the thermo-mechanical compressor and the results of the tests carried out using pure helium.

№	Name, unit	Symbol	Value
1.	Cylinder diameter, mm	D_C	106
2.	Displacement stroke, mm	L_C	96
3.	Dead volume, dm^3	V_E	0,78
4.	Relative dead volume, %	V_E / V_C	92
5.	One cycle period, s	τ_c	1,5…4,0
6.	Helium inlet pressure, MPa	P_S	7,45
7.	Heating water temperature at the inlet, K (°C)	t'_{H_2O}	302,6; (29,4)
8.	Average heater temperature, K (°C)	T_h	294,2; (21,1)
9.	Average cooler temperature, K (°C)	T_c	156,8; (-116,4)
10	Relative temperature	θ	1,88
11	Compression ratio	$\sigma = P_D / P_S$	1,2…1,3
12	Volumetric flow rate per 1 cycle at suction conditions, dm^3 per 1 cycle at P_0 = 0,1013 MPa, T_0 = 293 K, std. dm^3 per 1 cycle at P_0 = 0,1013 MPa, T_0 = 293 K, std. m^3/h	 f_S f_{0S} F_0	 0,078…0,183 10,0…23,5 9,0…56,4

CONCLUSIONS

The use of Internal heat regeneration processes in thermal compression systems allows the creation of sufficiently effective installations for the production of compressed gas - thermomechanical compressors.

The feature of the work performed is that two qualitatively different types of energy are consumed in installations for the production of compressed gas: thermal energy, directly used for gas compression, and mechanical energy, necessary to compensate for losses due to friction in seals, as well as to overcome hydraulic resistance of heat exchangers and gas pipelines. The principle of operation of such devices is based on cyclic changes in the temperature of the working fluid when in contact with warm and cold sources.

The most important result of the work is the development of a small-sized thermos-compressor with heat exchange processes provided by the introduction of the internal heat regeneration process between the heater and cooler through the built-in regenerator in the displacer.

In addition, the created low-temperature thermos-compressor differs from machines of the traditional type in several design and operational advantages. The drive based on permanent magnets eliminates the seal node and ensures the tightness of the working volume, preventing the penetration of oil and other by-products into the technological process, which is very significant for the inert gases production.

If there are high-boiling components in the pumped mixture, the minimum coolant temperature should be limited to avoid phase transitions (condensation or freezing). A promising option for this is a heat removal scheme using an intermediate coolant - krypton. Krypton is more preferable because it is safe and, at the same temperatures, its boiling pressure is 30...40% lower than, for example, that of methane.

This thermo-compressor can operate at pressures of P > 10 MPa and at a moderate compression ratio of 1.2...1.4, and also use the exergy of cryogenic product vapors, such as N_2, for driving.

At moderate coolant temperatures of T_c > 150 K, the maximum compression ratio is σ_{max} < 1.4.

In the case of compressing pure gases, such as helium, the coolant temperature can be reduced to T_c = 90 K. At the same time, the maximum compression ratio increases to σ_{max} < 1.8, and the performance under otherwise equal conditions increases by at least two times.

References

[1] Bush V. Apparatus for Compressing Gases. US Patent 2,157,229, 1939.

[2] Suslov A.D., Gorokhovsky G.A., Poltaraus V.B., Gorshkov A.M. *Kriogennye Gazovye Mashiny* [Cryogenic Gas Machines]. Moskva: Mashinostroenie, 1982. 213 p. (in Russian).

[3] Karabulut H. Thermodynamic Analysis of Bush Engine. *G.U. Journal of Science*, 2003, pp. 135-144.

[4] Blagin E., Dovgyallo A., Uglanov D. Numerical Modeling of Non-stationary Processes in Cryogenic Mechanical Thermo-Compressor. *International Journal of Mechanical Engineering and Robotics Research*, 2017, vol. 6, no. 4, pp. 258-262.

[5] Wu S.S., Wang J., Zhang H.C., Huang C.J. Thermodynamic Analysis of Ideal Thermo-Compressor Based on Euler View. *IOP Conference Series Materials Science and Engineering*, 2022, 1240(1):012035.

[6] Blagin E., Dovgyallo A., Uglanov D. Study of Different Factors Influence on Thermocompressor Performance. *International Conference on Mechanical, Aeronautical and Automotive Engineering*, 2017, vol. 108, 04001.

[7] Fischer F., Kuehl H.-D. Generation of CompressedAir by Cascaded Thermo-Compressors – Project Status. *19th International Stirling Engine Conference*, 2021, vol. 313, 04003.

[8] Sharifi N., Boroomand M., Kouhikamali R. Wet Steam Flow Energy Analysis Within Thermo-Compressors. *Energy*, 2012, vol. 47, is. 1, pp. 609-619.

[9] Ahmad K. Sleiti, Wahib A. Al-Ammari, Mohammed Al-Khawaja, Ahmad T. Saker. Experimental Investigation on the Performance of a Novel Thermo-Mechanical Refrigeration System Driven by an Expander-Compressor Unit. *Applied Thermal Engineering*, 2022, vol. 212, 118635.

[10] Fischer F., Kuehl H.-D. Analytical Model for an Overdriven Free-Displacer Thermo-Compressor. *Applied Thermal Engineering*, 2021, vol.185, 116251.

[11] Lin W.Y., Wu X.H., Yang J.L., Yang L.W. Experimental Study and Numerical Analysis of Thermo-Compressors with Annular Regenerators. *International Journal of Refrigeration*, 2013, vol. 36, is. 4, pp. 1376-1387.

[12] Ji W., Xue X., Wang J., Zhou Y., Chen L., Zhu W. Coupling Study of a Novel Thermo-Compressor Driven Pulse Tube Refrigerator. *Applied Thermal Engineering*, 2013, vol. 51, is. 1-2, pp. 630-634.
[13] Seth T., Eric J. Barth. Active Stirling Thermo-Compressor: Modelling and Effects of Controlled Displacer Motion Profile on Work Output. *Applied Energy*, 2022, vol. 327, 120084.
[14] Ahmad K. Sleiti, Mohammed Al-Khawaja, Wahib A. Al-Ammari. A Combined Thermo-Mechanical Refrigeration System with Isobaric Expander-Compressor Unit Powered by Low Grade Heat – Design and Analysis. *International Journal of Refrigeration*, 2020, vol.120, pp. 39-49.
[15] Sharifi N., Boroomand M. An Investigation of Thermo-Compressor Design by Analysis and Experiment: Part 1. Validation of the Numerical Method. *Energy Conversion and Management*, 2013, vol. 69, pp. 217-227.
[16] Sharifi N., Boroomand M. An Investigation of Thermo-Compressor Design by Analysis and Experiment: Part 2. Development of Design Method by Using Comprehensive Characteristic Curves. *Energy Conversion and Management*, 2013, vol. 69, pp. 228-237.
[17] Aristides M. Bonanos. Physical Modeling of Thermo-Compressor for Desalination Applications. *Desalination*, 2017, vol. 412, pp. 13-19.
[18] Kouhikamali R., Sharifi N. Experience of Modification of Thermo-Compressors in Multiple Effects Desalination Plants in Assaluyeh in IRAN. *Applied Thermal Engineering*, 2012, vol. 40, pp. 174-180.
[19] Noori Rahim Abadi S.M.A., Kouhikamali R., Atashkari K. Non-Equilibrium Condensation of Wet Steam Flow Within High-Pressure Thermo-Compressor. *Applied Thermal Engineering*, 2015, vol. 81, pp. 74-82.
[20] Wang J., Pan C., Luo K., Chen L., Wang J., Zhou Y. Thermal Analysis of Stirling Thermo-Compressor and Its Prospect to Drive Refrigerator by Using Natural Working Fluid. *Energy Conversion and Management*, 2018, vol. 177, pp. 280-291.

Information about the authors.

Simonenko Yury Mikhailovich. Doctor of Technical Sciences. Area of scientific interests: low-temperature systems for separation of multicomponent gas mixtures. E-mail: ysimonenko@cryoin.com

Chygrin Artem Aleksandrovich. Engineer-technologist. Area of scientific interests: low-temperature systems for separation of multicomponent gas mixtures. E-mail: achigrin@cryoin.com

Kostenko Evgeny Vladimirovich. Graduate student. Area of scientific interests: low-temperature systems for separation of multicomponent gas mixtures. E-mail: kostenko-zheka@hotmail.com